DESIGN OF A COMBUSTION CHAMBER
FOR AN EXPERIMENTAL GAS TURBINE

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APPROVED: ....

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LIST OF ABBREVIATIONS AND SYMBOLS

h .... Unit enthalpy, BTU per pound

v .... Velocity, feet per second

j .... Mechanical equivalent of heat, 778 foot pounds per BTU

g .... Dimensional constant, 32.2 feet per second per second

f .... Fuel-air ratio, pounds of fuel per pound of air

t .... Absolute temperature, ^{0}R

e_c .... Combustion efficiency, dimension less

h_c .... Chemical energy of fuel, BTU per pound

INTRODUCTION

PURPOSE

In December, 1946 the General Electric Co, published its bulletin No. DF 81523 on the performance, installation, and operation of an educational gas turbine, commonly called a bootstrap unit. This unit (like others which followed) did not produce any net work or thrust; the entire output of the turbine was absorbed by the compressor. Since then, there have been many similar units constructed both in industry and in engineering schools. The Georgia Institute of Technology started construction of its bootstrap unit in May 1948; the unit was ready for operation in April 1949. Unfortunately, it did not perform satisfactorily. Some modifications were tried; steam was injected just after the combustion chamber, and the starting air was left on during the operation of the unit. It was possible with these external cooling mediums to get it running, but only in a very narrow range of speeds without exceeding the temperature limits. Since the only reason for the units inability to operate properly rested in the design of the combustion chamber and compressor duct, it was decided to redesign these components.

It was the purpose of this paper to show why the new combustion chamber was designed as it was and to show how it performs relative to combustion chambers on jet engines and on other bootstrap units.

REVIEW OF THE LITERATURE

A search of the literature on combustion chamber design brings out clearly the fact that there are no analytically derived equations which correlate the various factors which affect combustion chamber performance. That is to say, one cannot find an equation, or set of equations, which will enable a designer to predict the effect on combustion chamber performance when one factor, say combustion chamber exit temperature, is varied and the others, say inlet air velocity, temperature and pressure, fuel characteristics, degree of atomization of the fuel, combustion chamber volume, size and arrangement of secondary air holes, are held constant.

The following are typical quotations from various papers on the subject of combustion chamber design:

(1) Walker[37] says, "It was, consequently, never possible to design and build a combustion chamber based on an intimate study of the combustion conditions likely to prevail, and on the chemical and physical characteristics of the fuel and air streams. For one thing, such fundamental data both as to the characteristics involved and as to the conditions of operation were partially, and in some cases, completely lacking. The greater part of the initial development was based on ad hoc experimentation and cut and try methods aided by a good deal of guessing."

(2) Nerad[24] states, "While the turbo jet combustor is an apparently simple device mechanically this appearance may be somewhat misleading because of the close interdependence of the various phases of its design. The hole and louvre arrangement, the fuel injection characteristics, the degree of uniformity of the air supply and the nature and location of the ignitor affect one another mutually. This complexity has had the result that a long program of cut and try development has been the usual rule in order to obtain highest performance in all respects from a given design."

(3) Redding[26] says, "At present, a remarkably small amount of fundamental understanding of the aviation gas turbine combustion chamber exists.... The great improvements in combustion chambers over the past few years have to a large extent resulted from careful testing under conditions simulating engine operation coupled with empirical modification."

(4) Shepard[30] states, "The development of combustion chambers to date has been accomplished by almost wholly empirical means. Although guided by a few broad principals, and it is as yet impossible to produce a combustion chamber from a set of design data without a considerable amount of experiment and trial and error modification."

It should be pointed out that although a tremendous amount of research work is being done on all phases of gas turbines, a great majority of it is classified. If it had been possible to obtain these classified papers, most of the apprehension associated with this project might have been avoided.

It should not be inferred from the foregoing that there is no information available on combustion chambers. Although it is true that the information which is available is of a most general character, certain trends were pointed out in practically all the pertinent literature. With these trends known and assuming that good guesses could be made when definite information was unavailable, the design of the combustion chamber was started.

APPARATUS AND INSTRUMENTATION

The main component of the gas turbine unit is a General Electric Type B-31 turbosupercharger (1)* which contains the turbine wheel, single stage centrifugal compressor, and a gear type lubricating oil pump.

The compressor duct (2) whose function is merely to lead the compressed air from compressor discharge to combustion chamber inlet has two sections, one being practically straight, the other is a 186° bend which has one continuous turning vane. The duct is 8" X 8" throughout except for a transition piece next to the compressor discharge. The transition piece is 5" X 7" on one end and 8" X 8" on the other; its axial length is about 12". It has 4 turning vanes at its compressor discharge end to reduce loss due to turbulence.

In the Combustion chamber (3) air and the fuel are mixed and burned, the products of combustion then going to the turbine.

The intake air duct (4) conducts ambient air through the cell block to the intake of the compressor. See reference 22 for brief discussion of the design of this component.

After the air goes through the intake duct, compressor, compressor duct, combustion chamber, and turbine, it is exhausted to the atmosphere by means of the exhaust duct (5).

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* Parenthesized numbers following name of part or component refer to corresponding or component part in Figures 5 through 11 inclusive.

A 3/4 hp., 220 volt, electric motor (6) vee belt connected to a centrifugal blower (7) supplies cooling air to the bearings of the supercharger and also to the turbine cooling cap. The blower is necessary because the supercharger is designed for operation at high altitudes in high speed aircraft and the bootstrap is both stationary and at approximately sea level. The colling effect of the air delivered by the blower must be substituted for the cold high velocity air going by the bearings when the supercharger is in normal operation.

A 110-15,000 volt transformer (8) supplies high voltage current to the spark plug.

Fuel is supplied with a war surplus aircraft pump (9) vee belt connected to a 1 hp., 110 volt electric motor (10). The pump can deliver up to 7 pounds of Diesel fuel per minute at pressures exceeding 1000 psi. Fuel is injected into the combustion chamber through a fixed orifice nozzle (11)*.

A Potentiometer Pyrometer (12) connected through a selector switch (13) to the six thermocouples is used for all temperature measurements.

Since the turbosupercharger is operated over a wide range of speeds, an electric tachometer set (14) is used. The indicator reads 1/500 rotor speed.

A beam type scale (15) is used for measuring the amount of fuel delivered to the unit.

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* Is similar to those used in domestic heating units.

Immediately before the 189° bend in the compressor duct there is a static pressure tap (16), impact tube (17), and thermocouple while at the entrance to the combustion chamber there is a static pressure tap (19) and thermocouple well (20). At the turbine inlet are the following: one static pressure tap (21), one impact tube (22) and two thermocouple wells (23), (24).

An aspirating thermocouple (25) is situated about three inches outside the turbine inlet duct. It is connected to the impact tube line by a tee connection. When a reading from this thermocouple is desired, the globe valve (26) which is in parallel with the manometer measuring total pressure at turbine inlet is opened. This by-passes the manometer and the temperature of the products of combustion passing the thermocouple can read.

A discussion as to the reasons for the type and positioning of the instruments will be given later.

DESIGN OF THE COMBUSTION CHAMBER

DESIGN OF THE OUTER DUCT AND TRANSITION PIECE

In Surine's paper[35], the most important conclusion drawn was that large volumes, low inlet air air velocities, and high air pressures were conducive to high combustion efficiency, low static pressure loss, and easy starting. With this fact in mind, it was decided to make the combustion chamber as large as possible. In keeping with the current design practice a combustion chamber of conical shape was decided upon. Since the length of the combustion chamber was limited (the compressor duct had already been designed, constructed, and installed by the writer) the only way to have a large volume was to have a large lateral dimension. Consideration of the working drawings of the compressor duct indicated that an outer duct with a maximum diameter of 21" while not being the largest possible would be the largest practical one. If the outer duct were made much larger it would be rather difficult to assemble and maintain the unit. The compressor duct being of square cross-section, it was necessary to build a transition piece from 8" X 8" to a 21" diameter. The axial length of the transition piece was quite arbitrarily set at 8" since it was thought advisable to have the maximum diameter close to the point of fuel injection. It had already been decided to inject fuel as far from the turbine inlet as possible (at the inlet to the transition piece) so as to give a maximum time for its vaporization, ignition, combustion and also for the mixing of the hot and cold gas streams. If the transition piece were made much longer, there would have been a possibility of the fuel droplets striking the flame tube wall and burning and/or evaporating and depositing carbon. The disadvantage of having such an abrupt change in cross-section is, obviously, that the "shock" loss is greater than if the change were more gradual. A good compromise between "shock" loss and the possibility of carbon deposition (not based on any calculations) seemed to be an 8" axial length.

The compressor duct had been designed so that the gases flowing through the combustion chamber would not have to turn any corners before getting to the turbine inlet section. After the transition piece was designed, constructed, and installed, measurements were taken from the inside diameter of the transition piece to the inside diameter of the turbine inlet section. The perpendicular distance between the centers of these parts was also measured. These measurements indicated that the center line of the turbine inlet section was coincident with that of the transition piece.

Since the turbine inlet section had a special flange whose mating piece was on the original combustion chamber, a 6 1/2" piece of the original combustion chamber containing this mating flange was cut off and used.

Consideration of the design of the compressor duct indicated that some provision had to be made for the expansion of the outer duct. The coefficient of thermal expansion of Inconel (of which the flame tube and outer duct was made) was found to be 6.4 X 10.6. The maximum temperature was estimated at about 2500° F. (this rather high temperature was chosen to afford a wide margin of safety.) The amount of expansion to be provided for was then easily obtained.

A simple expansion joint with only one pleat was decided upon since it, like the rest of the combustion chamber, was to be fabricated by the writer. There were no reasons other than pure intuition which dictated its design. During the brief trial runs of the bootstrap, the expansion joint seemed to operate satisfactorily.

It was necessary not only from a fabrication but also from an experimental standpoint that the transition piece and outer duct be two separate pieces which could be detached from each other with a minimum of trouble. Flanges of 1 3/4" radial width were welded to these two parts. The flanges were separated by a 1/16" asbestos gasket. Twenty four 5/16" cap screws through 3/8" holes were used for holding the parts rigidly together.

With the axial length and both diameters of the outer duct known as well as the length of the flange and expansion joint, the design of the outer duct was complete.

It should be pointed out at this time that the various components were not fabricated entirely from drawings. This would have been possible if the writer had had the services of competent machinists, sheet metal workers, and welders. As it was, when each part was made, it was fitted and then adjustments were made in the design of the other parts which followed.

Unfortunately, this was not possible in all cases. After part of the outer duct had been fabricated, it was found that its smaller end did not match the turbine inlet section. Instead, the center lines of these two parts were out of line by about 1 1/2". This was, no doubt, a result of distortion of the duct while it was being welded. The only way to remedy the situation without making a new outer duct was to cut out part of the compressor duct, fit the outer duct and flanged pieces properly, hold them rigidly in place, and then fit plates onto the compressor duct.

DESIGN OF THE FLAME TUBE

The flame tube in practically all combustion chambers in use in gas turbines or turbo-jet installations today have two distinct zones. The first is the reaction zone (also called the primary zone) where the fuel and part of the air delivered by the compressor are mixed and where combustion takes place. The second is the mixing zone wherein the products of combustion and the remainder of the air are combined; the mixture, cooled to a temperature dictated by metallurgical considerations then traveling to the turbine.

The reason for this separation and recombination of the gas streams is simply that the overall air-fuel ratio necessary to keep the turbine inlet temperature within limits is so high as to be practically incombustible. Vincent[36] says that the overall air-fuel ratio is usually about 125 pounds of air per per pound of fuel. When this is compared with air-fuel ratios in the combustible range (between 5 and 45 pounds of air per pound of fuel depending on the hydrocarbon being burned), the reason for the two separate zones becomes even more obvious. Furthermore, the length of the flame body (the region between the point of fuel injection and the point of complete combustion) is much longer if the separation of these zones is not adequately provided for. When sufficient combustion space is allowed, the extremely high temperature in the primary zone insures rapid and complete (usually 95-98%) combustion. Since the distance between the compressor duct and turbine inlet had to be sufficient for reasonably complete combustion, it was imperative to design the flame tube with a large primary zone and yet provide for sufficient space for a thorough mixing of the two streams. With these conflicting factors in mind and practically no engineering information available, the design of the flame tube was started.

As will be pointed out in subsequent paragraphs, the fuel-air ratio in the reaction zone of the flame tube is approximately the chemically correct one. It can easily be shown that a fuel-air mixture of this type will result in the products of combustion being at approximately 4500° F. This figure is, of course, dependent upon many factors, but can be considered to be close to being right for conditions which were present in the combustion chamber of the bootstrap unit. If there were a 2000° F temperature drop between the hot gas and flame tube wall, it (the flame tube wall enclosing the primary zone) would be at a temperature of 2500° F. Furthermore, the heat transferred by radiation from the gas might cause the temperature of the tube to be raised another 200 or 300° F. These extremely high temperatures, obviously indicate the need for a special alloy as material for the flame tube. Fortunately, it was possible to get a sheet of Inconel* for the fabrication of the flame tube and outer duct.

As indicated in the above, it was not possible to calculate the temperature of the flame tube. The reason is, quite obviously, because the condition (temperature, pressure and velocity) of the gas was unknown and also because information as to the factors affecting radiant transfer of heat were completely unknown.

*77% Nickel, 15% Chromium, 2% Iron, plus small amounts of copper, manganese, carbon and sulphur. Short time tensile strength at 2000°F is 11,000 psi.

It was hoped that the secondary air flowing over the outside surface of the flame tube would keep its temperature within reasonable limits. Furthermore, the only stress on the flame tube would be its own weight which was not expected to exceed 35 pounds. A reinforcing strap was welded around the periphery of the flame tube at the point of suspension to insure against mechanical failure. All indications pointed to the fact that the Inconel would be satisfactory for this application.

Way[39] suggests that the flame tube be designed so that the mixture in the primary zone be rich; he also suggests that a "suitable" coefficient be applied to the primary air orifice so that this rich mixture be obtained. Unfortunately, he does not say exactly how rich the mixture should be nor does he say what a suitable coefficient should be. A great majority of the other writers suggest that the mixture should be approximately the stoiometric amount and is often quite lean (20 to 30 pounds of air per pound of fuel).

Relying on the majority opinion, the writer decided to design the orifice so that the air admitted into the primary zone would be 110% of the chemically correct amount.

The General Electric Co.[12] in their Bulletin No.DF 81523 state that the compressor will deliver 200 pounds of atmospheric air per minute at sea level conditions when the compressor is rotating at 20,000 rpm and the discharge pressure is 32 psia. Assuming that the General Electric Co. defines ambient conditions as $80^{o}F$ and 14.7 psia and knowing the temperature of the compressed air, the average velocity of the discharge air was calculated to be 64 feet per second. Assuming further that the air velocity was constant over the cross-section of the duct, it was possible to predict how much air would be passing through an opening of a given size.

In order to determine the size of the primary air orifice, it was necessary to know how much fuel would be required to run the unit at 20,000 rpm. In the General Electric Bulletin it was stated that the turbine inlet temperature was 1100°F when the unit was operating at 20,000 rpm and not bleeding any air. Since the combustion chamber inlet and outlet temperatures were known and also the mass rate of flow of air, it was possible to predict the fuel requirements with reasonable accuracy. Calculation resulted in an approximate fuel requirement of 2.76 pounds per minute. 110% of the stochiometric amount of air would therefore be 45.9 pounds per minute. Knowing the velocity of the air entering the combustion chamber, the mass flow in the primary zone, and the density of the air, it was a simple matter to determine the ideal area of the primary air orifice. It came out to be 0.0918 square feet.

It was first thought that the coefficient to be used to arrive at the actual diameter of the orifice should be about 0.62, but after further consideration it was decided to use 0.8. This choice like many of the previous ones was completely arbitrary since there were no published data available for a situation which approximated this one. Using this coefficient, the diameter of the primary air orifice came out 5.13". This as rounded off to 5" since the actual velocity profile of the entering air was not expected to be a straight line but rather a curve with a maximum value at the center of the duct.

The next problem was to decide how many secondary air holes would be necessary and what their size and position would be.

Wells[40] does not feel qualified to make even the most general suggestion as to the design of the mixing section. He says, "The problem (of designing satisfactory combustion chambers) was difficult because the changes (in position and size of the secondary air holes) had to be made empirically. No theory was available for accurately predicting the effect of a change in the hole sizes in the liner wall". Any number of papers dealing with this part of the combustion chamber say substantially the same thing. The only one which makes any sort of concrete statement concerning the positioning and dimensioning of the secondary air holes is Shepard[31] who states that placing the holes in an in-line rather than in a staggered arrangement results in a better temperature profile at turbine inlet. He gives no reason for this but it can be explained by resorting to, what might be called, physical intuition. First, a statement as to the factors involved in the mixing process. The depth of penetration of an air jet into a hot gas stream will depend upon the relative magnitudes of the velocity and density of the streams, the diameter of the orifice through which the cold air is entering the hot gas region and the rate of heat transfer by conduction and connection between jet and main gas stream.

For the sake of convenience in comparison, let us assume that we have two identical combustion chambers one of whose mixing section is made up of a number of lateral holes in a scattered arrangement, the other of whose lateral holes are made in an in-line arrangement. Let us assume further that the same quantity of cool air is being supplied to both and that we will take temperature profiles from both at the same axial distance from the first mixing hole. To further simplify the discussion let us also assume that the diameters of the various holes in both combustion chambers are all the same.

As the main gas stream passes the first hole of the mixing section, the air jet in both combustion chambers will have the same depth of penetration and the mixture immediately behind the first sir hole will, in both cases, be identical. In the in-line arrangement, the next air jet will have a greater depth of penetration since the gases that flow past it are coller and are not travelling as fast as the gas stream flowing past the first air hole. Thus if we have sufficient space, it would be possible for this arrangement of air holes to actually separate the main gas stream into hot sections with a cool one between them. Now let us look at what is happening to the streams in the combustion chamber with the scattered air hole arrangement. As previously stated, the mixture behind the first hole would be the same as the in-line arrangement. But instead of following up this first quenching with still another, we allow another air jet angularly displaced from the first to discharge into the still uncooled main gas stream. Its penetration is the same as the first. Thus after the main gas stream travelled a certain distance downstream, the only portion of the stream that is cooled is the one closest to the flame tube wall. Remembering that the temperature profile of the gas stream before the mixing section is similar to the velocity profile of a fluid in turbulent flow, we see that we have cooled that portion of the stream which was cooler to begin with. Naturally, after the main gas stream has travelled through the mixing zone a considerable portion of it has been cooled but the center core (which is hottest) has not. In a word, the staggered arrangement has allowed us to cool the periphery of the main gas stream, while the in-line arrangement has allowed us to penetrate into the hottest part of the stream. This penetration will, therefore, result in a more uniform temperature profile after mixing. With the above in mind, it was decided to have the secondary air holes in an in-line arrangement; their location and size still had to be determined.

With no engineering data available to determine the total secondary air hole area, another guess was made: the cross-sectional area of the compressor duct was 64 sq. in., the cross-sectional area of the primary air orifice was 13.2 sq. in. The latter was subtracted from the former and the remainder was 50.8 sq. in. This was assumed to represent the total ideal secondary air hole area necessary. Applying a coefficient of 0.8, the total secondary air hole area came out to be 63.6 sq. in. Having the rows of air holes spaced at 450 intervals around the periphery of the tube gave eight holes per cross-section. Arbitrarily deciding to make all the secondary air holes 1 $\frac{1}{4}$" diameter, it was found necessary to have six holes per row or a total of 48, 1 $\frac{1}{4}$" holes. This gives a total secondary air hole area of 58.9 sq.in. which was considered close enough to the amount needed.

It was now necessary to determine exactly where these holes would be. Mock,[22] Shepard,[29] LLoyd[18] and Hawthorne[15] emphasize the fact that if the secondary air holes are placed so that they discharge cold air into that region wherein the process of combustion is taking place, the cold secondary air will quench part of the flames below the ignition temperature and halt the combustion process before it has gone to completion. This, obviously, would have a definite adverse effect upon combustion efficiency. Furthermore, they say that the flame body will be increased if sufficient combustion volume is not provided.

Another guess was made: the edge of the first secondary air hole would be 18" from the primary air orifice. The holes would be equally spaced from that point to the end of the flame tube.

It was further decided to make the tube $\frac{1}{2}$" shorter than the outer duct to allow for expansion. Unfortunately, the distortion caused by welding the flame tube would not allow it to fit the outer duct properly. It was necessary to cut off about two inches from the turbine end of the tube before the parts mated as they should. This made the flame tube 38" long.

The maximum diameter of the flame tube and the maximum of the outer duct were to be at the same axial distance from the compressor duct. The maximum diameter of the flame tube was determined by finding the ratios of the flame tubes and outer ducts of the jet engines in the Mechanical Engineering Laboratories. The ratio in both cases was 3:4. Applying this ratio, the diameter of the flame tube for the boot strap came out 15.5". The reason for taking this ratio as design information was, as in previous cases, that no other information was available.

The design of the flame tube was then complete.

FUEL AND IGNITION SYSTEMS

Neither the fuel nor the ignition systems were, in the true sense of the word, designed, Rather they were assembled from equipment which was available. This equipment was modified to a greater or lesser extent to fit the particular requirements of the bootstrap unit.

The fuel nozzle obtained was the same as used in domestic oil-burning units. The modification was merely the enlargement of the nozzle grooves so that it would pass 3 pounds of fuel per minute at a pressure of 1000 psi. After the first set of trial runs were made, it was found that the actual fuel requirements were larger than anticipated. Since it was impossible to increase the fuel pressure and still have a reliable unit (the vee belt connecting the pump and electric motor would start to alternately slip and hold when fuel pressure exceeded 1200 psi), it was necessary to increase the size of the grooves. Since the exact fuel requirements were unknown, the grooves were purposely made quite large. When the unit was reassembled, it was found that the nozzle would pass sufficient fuel to run the unit at 20,000 rpm when the fuel pressure was about 500 psi.

A gear type fuel pump was obtained; no modifications were necessary to fit it directly into the proposed fuel system.

In view of the fact that the fuel tank was to be unenclosed, a fuel filter was obtained and installed without any modifications.

The simplest and most reliable means of varying the fuel pressure was by means of a parallel bleed-off line. Control was to be exercised by means of a right angle needle valve on the parallel line. The control system proved to be particularly satisfactory because pressure could, in case of emergency, be reduced from 1000 psi to 0 psi by turning the control valve handle through a very small angle (about 60°). A fuel shut off valve was installed just after the pressure gauge because when the gauge read 0 psi there was just enough actual pressure in the line to make the nozzle drip. By closing the fuel shut-off valve positive stoppage of flow to the nozzle was insured.

Since the velocity of flame propagation on the type of fuel-air mixture which was to be encountered in the flame tube was about 2-3 fps[31] and the velocity of the air entering the primary zone of the flame tube was about 64 fps, some mechanism to either decelerate or reverse the air stream was absolutely essential to the establishment of a zone wherein the combustion process could take place. This mechanism was to be the flame holder.

Mock[23] suggests a great many different methods to create a stabilized flame front ranging from an ordinary disc placed immediately before the point of fuel injection to an elaborate honeycomb of orifices and passages for the air around the fuel nozzle.

In order to keep the fabrication problem within reason and still be certain that the flame holder would function properly, one of the simpler designs suggested by Mock was decided upon: a frustrum of a cone with fuel injection at the smaller end and the larger end facing downstream. Unfortunately, Mock made no recommendations as to dimensions for any of the types of flame holders he mentioned.

The larger diameter of the cone had to be smaller than the primary air orifice (since the entire flame holder was to be inside the flame tube) and was arbitrarily chosen to be $4\frac{1}{2}$". The smaller diameter, since the flame holder was to be welded to the fuel nozzle adapter, was necessarily 3/4" in diameter. The length was determined by deciding to have the area between the edge of the flame holder and the flame tube wall equal to the area of the primary air orifice. The axial length of the cone came out to be 1 1/8".

The unit was disassembled after the first set of trial runs, and the flame holder was inspected. A coat of greasy soot about 3/32" thick was found completely covering it. It was assumed that the flow reversal caused by the flame holder was so pronounced that burning fuel was being thrown on its face, the fuel then cooling below a temperature where complete combustion could take place. In order to eliminate this, many small holes (0.045" diameter) were drilled in the flame holder surface, the holes being on $\frac{1}{2}$" centers. This would cause small streams of air to encounter the fuel-air mixture as it was just about to hit the face of the flame holder and thus prevent the deposition of carbon. The system seemed to work fairly satisfactorily, since only a very thin layer of powdery carbon was deposited on the flame holder after the second set of trial runs.

The ignition system, while it was a small component of the bootstrap unit and in operation only a small fraction of the time that the unit was running, was so important that as much time was spent on it as on the entire flame tube assembly.

The spark plug was salvaged from the General Electric jet engine in the Mechanical Engineering Laboratory. Since the spark gap was only about 3/32", the low tension part of the plug was discarded and a low tension point was welded to the flame holder. The spark gap was adjusted to 5/16". As a mounting the spark plug holder from the jet unit was bolted to the flame holder.

The reason for getting the 110-15,000 volt transformer was that it was the only high voltage transformer available.

The ignition system worked excellently; during the trial runs, combustion started to take place as soon as the fuel shut off valve was partially open.

MISCELLANEOUS

The method of flame tube support, since it was a minor detail, was arrived at without any calculations for stress. The only important factor to be considered was that of assembly and maintenance. It was decided that the flame tube would have one main support to hold it vertically and that spacers would be used to hold it in the correct horizontal position. Two vertical strips of Inconel each approximately 3/16" thick were to be welded to the inside of the outer duct at its uppermost point. The flame tube was to have one strip of the same size welded to its uppermost point, and this strip was to fit in between the two on the outer duct. Two 5/16" stainless steel bolts were passed through the three strips and held them rigidly together. Because of fabrication difficulties, it was necessary to weld the two strips for the outer duct to a third one so that the cross-section of the sub-assembly resembled the Greek letter "pi". The entire subassembly was then welded to the inside of the outer duct.

The spacers, since they would not be subjected to any appreciable mechanical stress, were to be made of 1" X 1/8" black iron strip. Their length was determined by placing the flame tube inside the outer duct as it would be in service and fitting the spacers in between the outer duct and flame tube. They were welded to the outer duct in that position.

Since the drain had to connect the flame tube and a valve on the outside of the combustion chamber and still allow the flame tube and outer duct to be separated easily, it was decided to place a union between them. All connections were to be welded and $\frac{1}{4}$" pipe and fittings used throughout.

CONCLUSIONS

RESULTS

After the completion of the design and construction of the combustion chamber, it was installed, and test runs were made. As has been previously mentioned, after the first set of trial runs, the fuel and ignition systems were modified. It was possible, after these modifications were made, to get the unit running at 20,000 rpm. Unfortunately, the combustion chamber did not perform as well as had been expected. The writer had been of the opinion that this combustion chamber would enable the unit to operate at 20,000 rpm at a turbine inlet temperature between 900-1100^{o}F. This opinion was based on the fact that the original bootstrap unit built by the General Electric Co. in 1946 operated at a turbine inlet temperature of approximately 1100^{o}F. The combustion chamber on this unit was the same as used in early models of a General Electric turbo-jet engine. It was assumed by the writer that the velocity of the air in this combustion chamber would be higher than in the new one at the Georgia Institute of Technology since the General Electric combustor was designed for a unit whose size and weight had to be kept at an absolute minimum. With air velocity higher, the static pressure drop must also be higher; it was, therefore, reasonable to assume that the new combustion chamber on the bootstrap at the Georgia Institute of Technology would perform as well, if not better, than the combustion chamber on the original

General Electric bootstrap unit. This better performance, it was further assumed, would enable the Georgia Institute of Technology bootstrap unit to run at a lower temperature than General Electric's (at any given speed). This was not the case; the turbine inlet temperature of the bootstrap at the Georgia Institute of Technology was 1530°F at a speed of 20,000 rpm. Hughes[17] states that the bootstrap developed by Allis-Chalmers Manufacturing Co. (which used two combustion chambers in parallel) operated at a turbine inlet temperature of 1250°F at 20,000 rpm.

With the comparatively mediocre performance of the new Georgia Institute of Technology bootstrap in mind, the performance of the original combustion chamber on the Georgia Institute of Technology bootstrap will be cited. As mentioned in the introduction to this paper, the original bootstrap unit could not be operated without external cooling mediums (steam injection at the turbine end of the flame tube, and utilization of the starting air after combustion was established). With these, it was possible to run the unit at 1400°F at a speed of 14,000 rpm. Stable operation could not be obtained at any other speeds without exceeding the 1400°F limit at turbine inlet. Before the design of the second combustion chamber was started the writer modified the fuel system on the original unit and was able to run it at speeds from 6,000 to 20,000 rpm at turbine inlet temperatures ranging from approximately 1200° to 1350°F respectively. In all fairness to the designer of the original combustion chamber, it must be pointed out that the bootstrap was not equipped with an intake air duct at the time these runs were made. The unit would have performed slightly more satisfactorily had this duct been installed.

Aside from the fact that the new combustion chamber allowed the bootstrap to operate at a lower inlet temperature, there is another fact that should be mentioned. It was observed when the original bootstrap at the Georgia Institute of Technology was running at low speeds (6,000-12,000 rpm), that flames were passing the turbine wheel. At speeds exceeding 12,000 rpm, flames could be seen inside the exhaust duct. At 20,000 rpm, it was noticed that burning was still taking place at a point about 3 feet past the end of the exhaust duct. Furthermore, when a rapid acceleration was caused by literally jamming fuel into the combustion chamber, the fuel was still burning at a point 10 to 12 feet past the exhaust duct. As soon as the acceleration was stopped, the flames subsided to the above-mentioned point three feet past the exhaust duct. As will be noticed from the photographs, the end of the exhaust duct is about 8 feet from the turbine inlet section. All indications point to the fact that a great deal of the combustion was taking place outside the combustion chamber. This fact was further verified when temperature measurements were taken just after the flame tube and and at turbine inlet. The temperature just after the flame tube was about 1230°F and at turbine inlet about 1850°F at 20,000 rpm.

If the chamber had been performing properly, the temperature just after the flame tube would have been higher than at turbine inlet. In addition, since burning was taking place past the turbine inlet, the temperature indicated by the thermocouple at that point did not represent nozzle box temperature, but was lower by an amount depending upon how much combustion had taken place between it and the nozzle box exit. This situation was brought about by the fact that the secondary air holes were too close to the point of fuel injection.

In contrast to this, the only visible sign that combustion was taking place in the bootstrap unit with the new combustion chamber was that the turbine blades and nozzle box were cherry red (when observed from a safe distance). No smoke was observed coming out of the exhaust duct or past the turbine wheel even at 20,000 rpm. It was possible to see the path that the exhaust gases were taking only because of the difference in the index of refraction between exhaust gases and the ambient air.

Ayers[1] et al report that the bootstrap unit at Purdue University required water injection to limit turbine inlet temperature to 1600°F. In contrast to the situation with the original bootstrap at the Georgia Institute of Technology, where there were no instruments with which to measure the amount of steam injected into the combustion chamber, the bootstrap unit at Purdue University had an elaborate set of instruments which enabled the operators to make all important measurements. When the Purdue University bootstrap was operating at a speed of 12,000 rpm, 350 pounds of water per hour were required to limit turbine inlet temperature to 1600°F; at 15,000 rpm, 650 pounds per hour were required; and at 18,000 rpm, 950 pounds per hour. When the curve (in this case, a straight line) of water requirements vs speed was extrapolated to 20,000 rpm (it wasn't possible to get the unit running at 20,000 rpm because the water system capacity was insufficient) the estimated water required came out to be approximately 1140 pounds per hour - hourly requirements of over one-half ton of water.

After consideration of the preceding paragraphs, the performance of the present combustion chamber on the bootstrap at the Georgia Institute of Technology does not show up too poorly.

Unfortunately, it was not possible to get exact quantitative data on the performance of the combustion chamber. This extremely regretable situation was brought about by a series of equally regretable mishaps. The first occured shortly after the writer started working on the original bootstrap. As previously mentioned, the original unit required the use of steam to limit turbine inlet temperature. On one occasion, the writer forgot to turn the steam on; when this was discovered, the pyrometer indicated a temperature of over 2000°F. The unit had just been started and, therefore, its speed was only about 9,000 rpm; nevertheless, temperatures of this magnitude shortened the life of the turbine blades. In addition, the unit was (for various reasons) brought up to 20,000 rpm at a turbine inlet temperature of about 1850°F many times. Since the turbine blades were not designed to withstand high stresses at such extreme temperatures, it was inevitable that they should fail. Besides this, since the starting air, of necessity, was left on, the blades were subjected to severe heating and quenching cycles. This resulted in accelerating the failure of some of the turbine blades by effecting their crystal structure. Six blades failed on the original turbosupercharger in February, 1951. The second turbosupercharger was installed, but the unit was run only once before the original duct and combustion chamber were scrapped.

After the new combustion chamber was installed, the unit was run about six times before the second mishap occurred. The unit had been running at 20,000 rpm for some time when a sheet of flame was observed coming from the section between the compressor diffuser and turbine nozzle box. The unit was immediately shut down, and the fire extinguished. Investigation proved that the bearing next to the turbine wheel had been leaking oil badly. It had leaked past the bearing, poured over the hot nozzle box, and then flashed into flames. Damage was negligible in that only a few thermocouple leads were burned. A most unfortunate situation then presented itself. It was that the turbosupercharger could not be considered safe enough to run, and it was impossible to obtain another one because of the national emergency. There was only one alternative, and that was to consider the thesis complete because the combustion chamber had performed fairly well, and also because it was impossible to continue further development work.

PROPOSED TEST PROCEDURE AND CALCULATIONS

The writer had intended making runs and taking data necessary to arrive at percent static pressure drop across the combustion chamber and combustion efficiency, since these are the two most important items in the evaluation of a combustion chamber. The others (reliability under mechanical stress, vibration, and high temperatures, ease of ignition, and temperature profile at turbine inlet) would have been difficult to measure and would have required the expenditure of considerable money to instrument the experiments properly.

The first item, percent static pressure drop, could easily have been calculated after measuring the static pressure at inlet and outlet of the combustion chamber, pressure taps Nos.19 and 21;* subtracting inlet from outlet static pressure and dividing the remainder by inlet pressure would give this item immediately.

The calculation for combustion efficiency would have been slightly more involved. Vincent[36] derived a simple equation starting with the steady flow form of the First Law of Thermodynamics. Vincent's equation reduced to:

$$h_2 + \frac{v_2^2}{2gJ} + f)0.5T - 375 + e_c h_c) - 1 + f(h_3 + \frac{v_3^2}{2gJ}) \qquad (1)$$

It was to be assumed that the enthalpy of the air was a pure function of temperature (this assumption was permissable because of the low pressures, 32 psia, and the high temperatures, 1500°F, involved). The enthalpy terms in the above equation could then be evaluated by

*Refer to Figure 5 through 11 inclusive.

temperature measurements before and after the combustion chamber; thermocouples Nos.20 and 24 in this case. The velocity terms were to be evaluated by measuring the static pressure, impact pressure, and temperature before and after the combustion chamber. Because of the 189° bend before the combustion chamber inlet, it was thought advisable to measure these quantities (for the inlet section of the combustion chamber) at the downstream end of the straight section of the compressor duct; pressure taps Nos.16 and 17, and thermocouple No.18 were to be used. The velocity obtained by calculation based on these measurements was to be corrected to conditions existing at the combustion chamber inlet by measuring the temperature and pressure there (pressure tap No.19 and thermocouple No.20) and assuming incompressible flow throughout the compressor duct. The velocity at turbine inlet was to be calculated based on similar measurements using pressure taps No.21 and 22 and thermocouple No.23. No correction would have been needed in this case. Naturally, it would have been necessary to have applied suitable coefficients to the velocities calculated in each instance so that the average velocity was used and not the maximum velocity. Because of the extremely high Reynolds number encountered (1,000,000 in the compressor duct and even higher in the combustion chamber) a coefficient of about 0.9 was to be used.

The mass flow of air was to be calculated by using the average velocity in the compressor duct and the cross-sectional area of same. The measurement of the fuel supplied was to have been obtained directly by weighing. The ratio of mass rate of flow of air and the mass rate of fuel supply was the "f" term in equation (1).

The chemical energy, h_c of the fuel supplied was known to be 19,025 BTU per pound, and was to be directly applied to the equation.

Every quantity except e_c, combustion efficiency, was then known; it could have been easily solved for.

The writer had intended to make calculations of combustion efficiency and percent static pressure loss with the unit running unthrottled at varying speeds and also with it running at constant turbine inlet temperature at varying speed. The latter runs would simply have required the use of a damper in the air intake duct.

It was also intended to plot turbine inlet temperature vs rotor speed. Naturally, no calculations would have been required except, possibly, averaging the temperature reading obtained.

The writer was confident that the combustion efficiency curve would have been very close to a straight line at about 97% under all conditions. The percent static pressure loss would have been a straight line from the origin with a maximum value of about 4% at 20,000 rpm.

RECOMMENDATIONS

Obviously, the first and most important step to be taken is to replace the turbosupercharger. It is the writer's opinion that if the turbosupercharger which is installed in place of the present one is a used one, something similar to the two accidents mentioned previously will happen in a very short time after installation. It would be pure folly to think that a used turbosupercharger will render acceptable service by the very fact that it has been removed from its installation. The present supercharger should be replaced only by a new one.

The combustion chamber was disassembled after the fire. Inspection of the flame tube indicated that some changes in the design must be made. For ont thing, there are signs of intergrannular corrosion in a region extending from the maximum diameter to about 12" downstream from this point. It should be emphasized that the corrosion is not serious, but it does indicate that the flame tube wall has reached very high temperatures. In order to avoid failure due to intergrannular corrosion, it is suggested that the primary air orifice be enlarged so that the temperature in the primary zone be reduced. As has been mentioned previously, the flame body will be lengthened by a decrease of the primary zone temperature. The optimum size of the primary air zone with consideration of the flame body length must be arrived at by purely cut and try methods.

The enlargement of the primary air orifice is highly desirable from another standpoint. Consideration of Figure 4 shows that the secondary air must follow a path which offers a relatively high resistance to its flow. When it is remembered that approsimately three times more air must follow this high resistance path than that which goes through the low resistance path (that of the primary air), a reduction in the amount through the high resistance path will result in a lower static pressure loss across the combustion chamber. With a lower static pressure loss, the unit will run at a lower turbine inlet temperature at any given speed.

It is believed by the writer that a smaller flame holder would create sufficient flow reversal to stabilize the flame front. A reduction in the size of this part would reduce the amount of turbulence in the flame tube and result in a decrease in the static pressure loss across the combustion chamber. Unfortunately, no recommendation as to optimum size can be given. Cut and try methods must be employed to arrive at a suitable diameter.

A design change which should be made as soon as possible involves the bend in the compressor duct. This design change is necessary because the writer failed to consider the maintenance and adjustments of the fuel and ignition systems during the initial design. In order to make any adjustments in either of these systems (which heretofore have proved to be most troublesome), it is necessary to remove the entire combustion chamber from the unit. In view of the fact that this operation requires the services of at least three and preferably four men, a modification which would enable one man to do the job is highly desirable. This could be accomplished if the bend in the compressor duct were cut in half transverely. That is to say, that if the 189° bend were two separate 94.5° bends, the entire fuel and ignition system could be removed without distrubing the combustion chamber itself.

Since this modification can be made with a minimum of trouble and is so important, it should be done even if the turbosupercharger cannot be replaced immediately.

The fuel nozzle should be modified. When the writer enlarged the grooves on the pintle before the second set of trial runs, he made them too big. As a result, when the fuel pressure is only about 500 psi, the nozzle is passing sufficient fuel to run the unit at 20,000 rpm. To utilize the capacity of the pump as much as possible, it is suggested that the nozzle grooves be reduced in size so that they will pass all the necessary fuel at 1000 psi. After the fuel nozzle has been modified, it might be that the unit cannot be run at low speeds (4,000 - 3,000 rpm). This might happen if the grooves are comparatively large. If this is the case, the writer suggests that two different nozzles be used, one of high speed, the other for low. If the modification of the compressor duct is made, the nozzles could be changed with a minimum of trouble.

The bearing blower unit should be tested to see if it is delivering about 200 cfm at 6" water total pressure. If it is not, replace it with a unit that will.

The intake air duct should be tested to see if it actually does help the unit run cooler. If there is no appreciable difference in turbine inlet temperature with and without the duct, then its straight section should be modified into a venturi meter. The entire duct should then be calibrated. The reason for this is, naturally, so that the calculation for mass flow of air based on total pressure in the compressor duct can be checked.

Since the thermocouples at the turbine inlet section (with the exception of the aspirating one) are exposed to direct radiation from the primary zone of the flame tube, they must indicate a temperature higher than that of the gas passing them. When it is remmebered that the temperature at this point is a critical item not only in the operation of the unit, but also in any experiments that can be conducted on the combustion chamber or any other component, it is of the utmost importance to be able to measure as close to the actual gas temperature as possible. This can be done best with a shielded thermocouple; therefore, this piece of equipment should be obtained.

It is the writer's opinion that the transformer should be placed outside the cell block because of the high air temperatures that are reached inside when the unit is running.

All the thermocouple leads should be arranged so that they do not span the turbosupercharger. Radiation from the nozzle box causes deterioration of their insulation.

A large CO_2 fire extinguisher should be placed close to the unit.

At least one more hole should be made through the top of the cell block because of the large number of pressure lines which will eventually run from the unit to the manometer board.

At least two more manometers should be made and installed, and all pressure taps should be piped to the manometers.

After all or some of the above modifications and additions are made, the combustion chamber performance should be evaluated.

With the addition of some more equipment, the performance of the bootstrap unit could be improved and various factors effecting complete gas turbine units could be determined. The first is the installation of a high pressure water injection system at the compressor intake and in the combustion chamber. With the intake air already a variable, the addition of this water injection system would enable the operator to vary conditions throughout the unit almost at will.

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APPFNDIX I

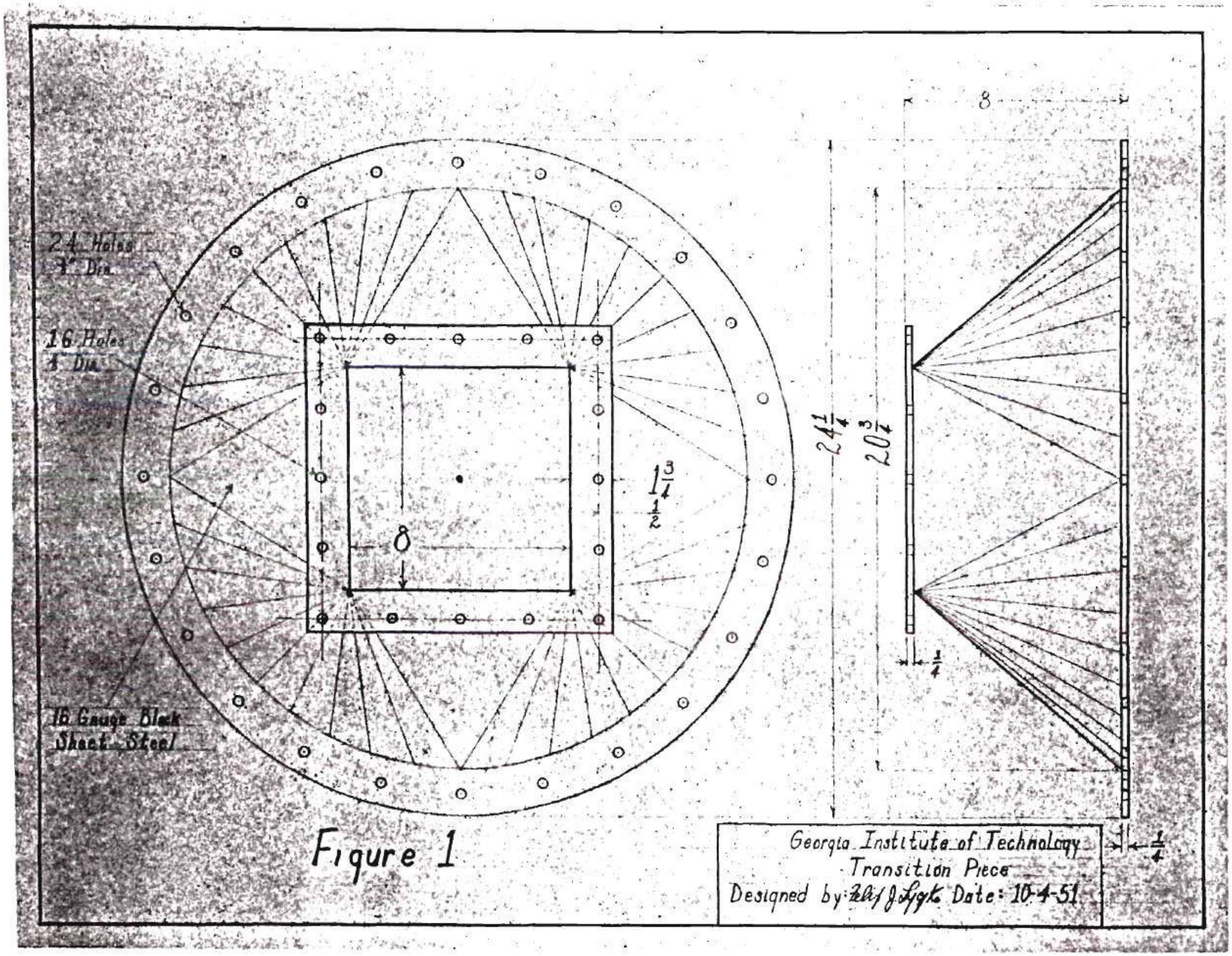

24 Holes
Dia
16 Holes
Dia
16 Gauge Black
Sheet Steel
8
$1\frac{3}{4}$
$\frac{1}{2}$
8
$24\frac{1}{4}$
$20\frac{3}{4}$
$\frac{1}{4}$
$\frac{1}{4}$
Georgia Institute of Technology
Transition Piece
Designed by
Date: 10-4-51

Figure 1

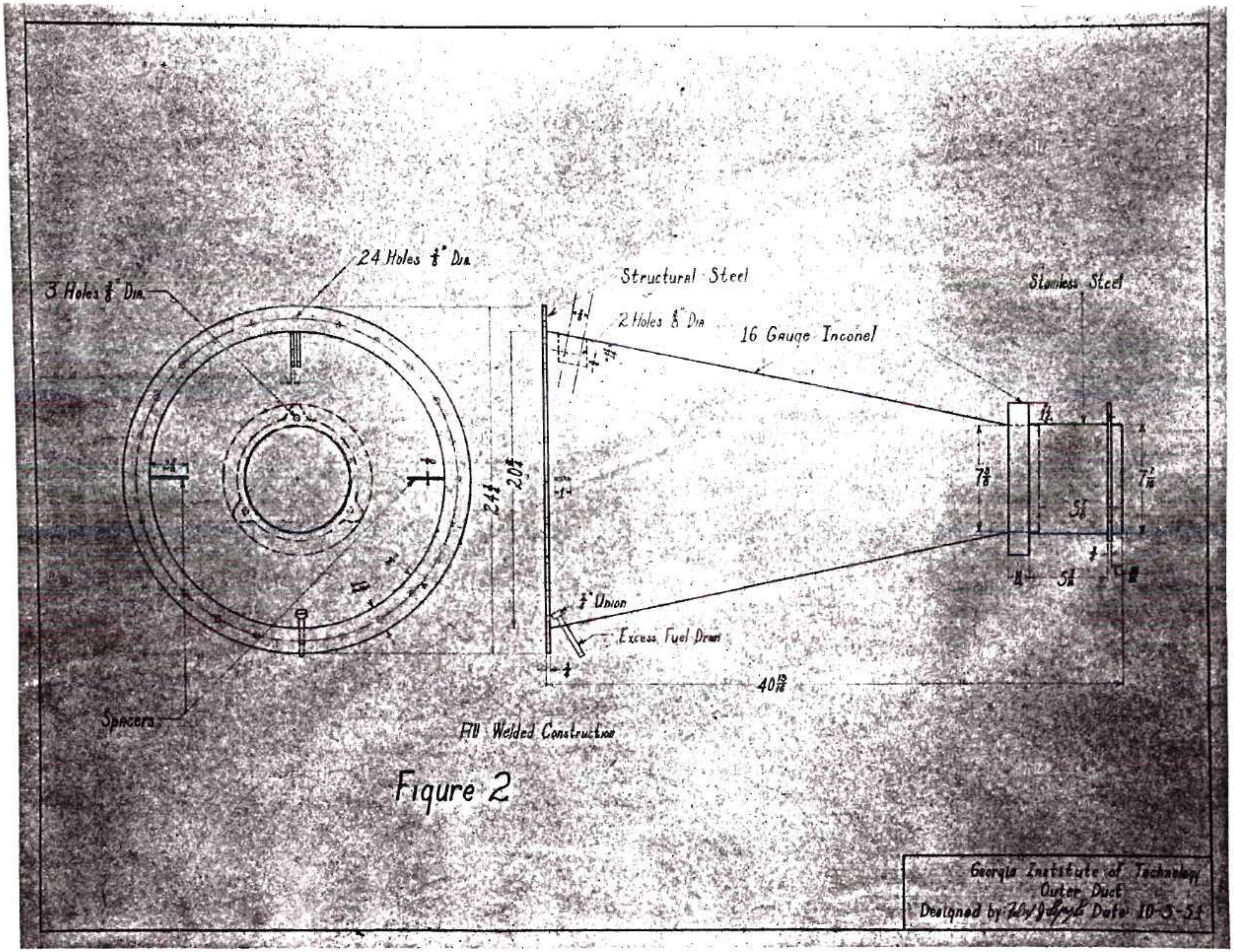

24 Holes
3 Holes
Structural Steel
2 Holes
16 Gauge Inconel
Stainless Steel
Union
Excess Fuel Drain
Spacers
All Welded Construction
Georgia Institute of Technology
Outer Duct
Designed by
Date: 10-3-51

Figure 2

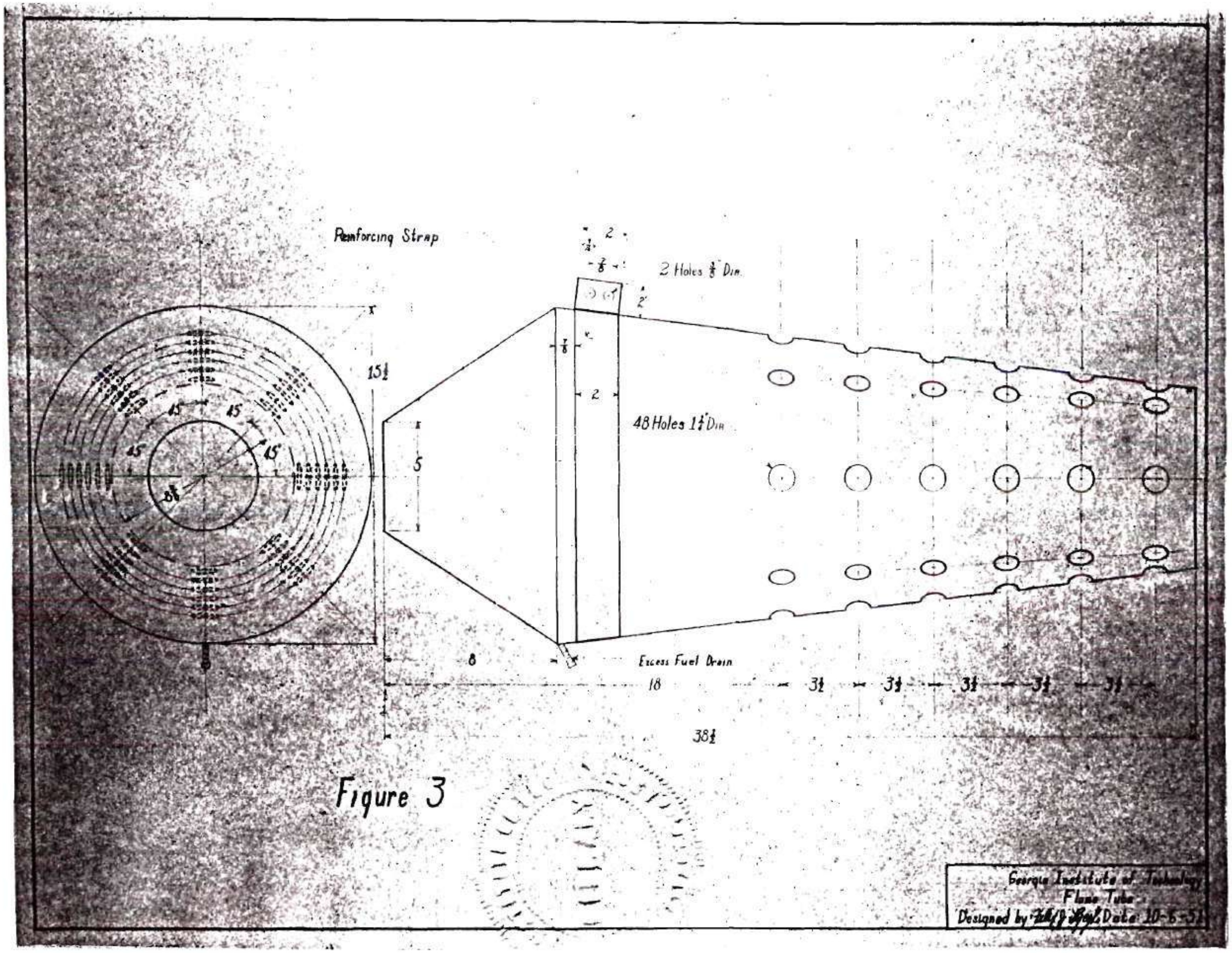

Reinforcing Strap
2
2 Holes $\frac{3}{8}$ Dia.
48 Holes $1\frac{1}{4}$ Dia.
$15\frac{1}{2}$
45°
$8\frac{5}{8}$
5
8
Excess Fuel Drain
18
$3\frac{1}{2}$
$38\frac{1}{2}$
Georgia Institute of Technology
Flame Tube
Designed by
Date 10-6-51

Figure 3

49

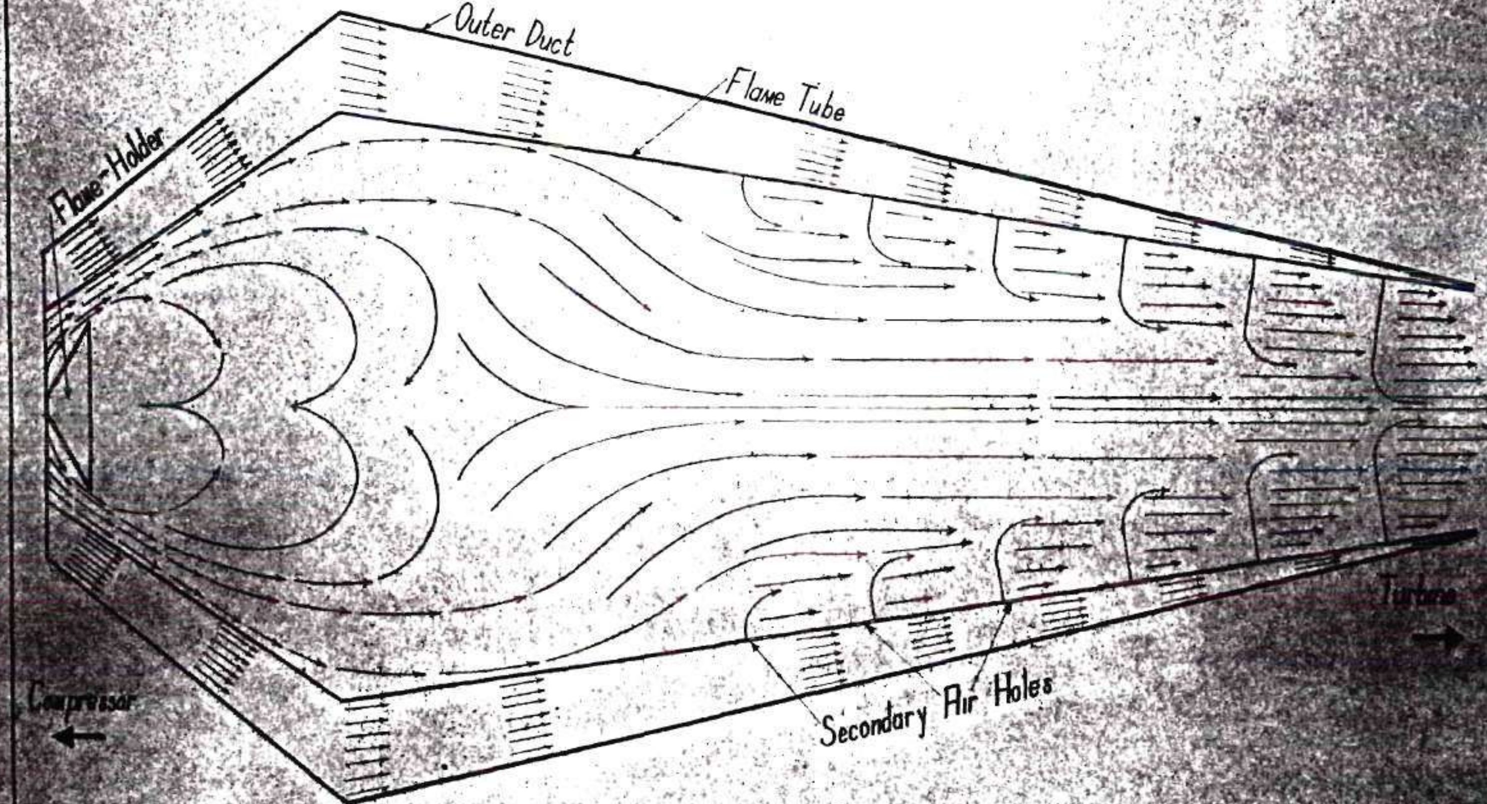


Figure 4

Georgia Institute of Technology
Diagram of Idealized Air-Flow Pattern
in New Combustion Chamber
Drawn by: Felix J. [signature] Date: 10-1-51

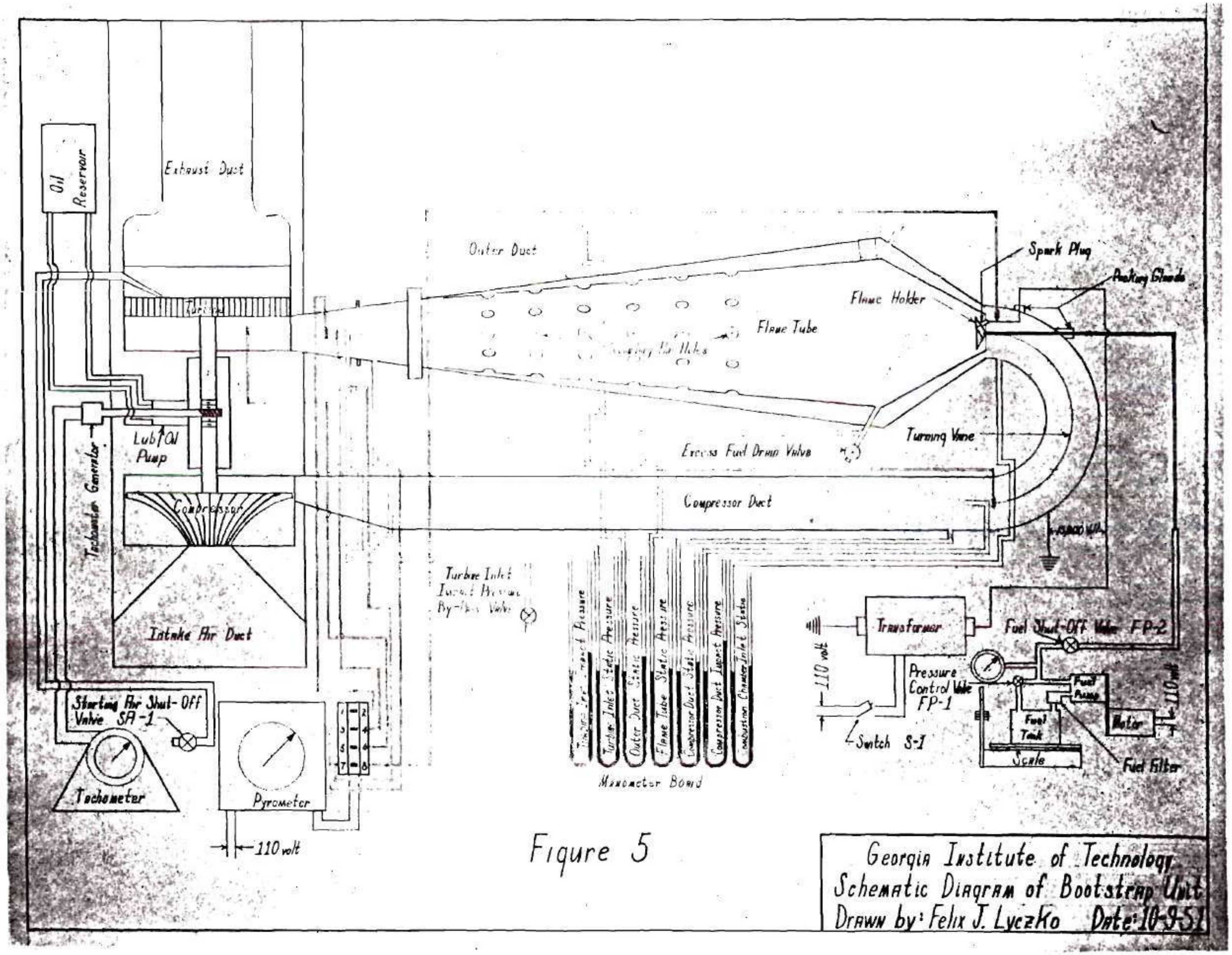


Figure 5

Georgia Institute of Technology
Schematic Diagram of Bootstrap Unit
Drawn by: Felix J. Lyczko Date: 10-9-51

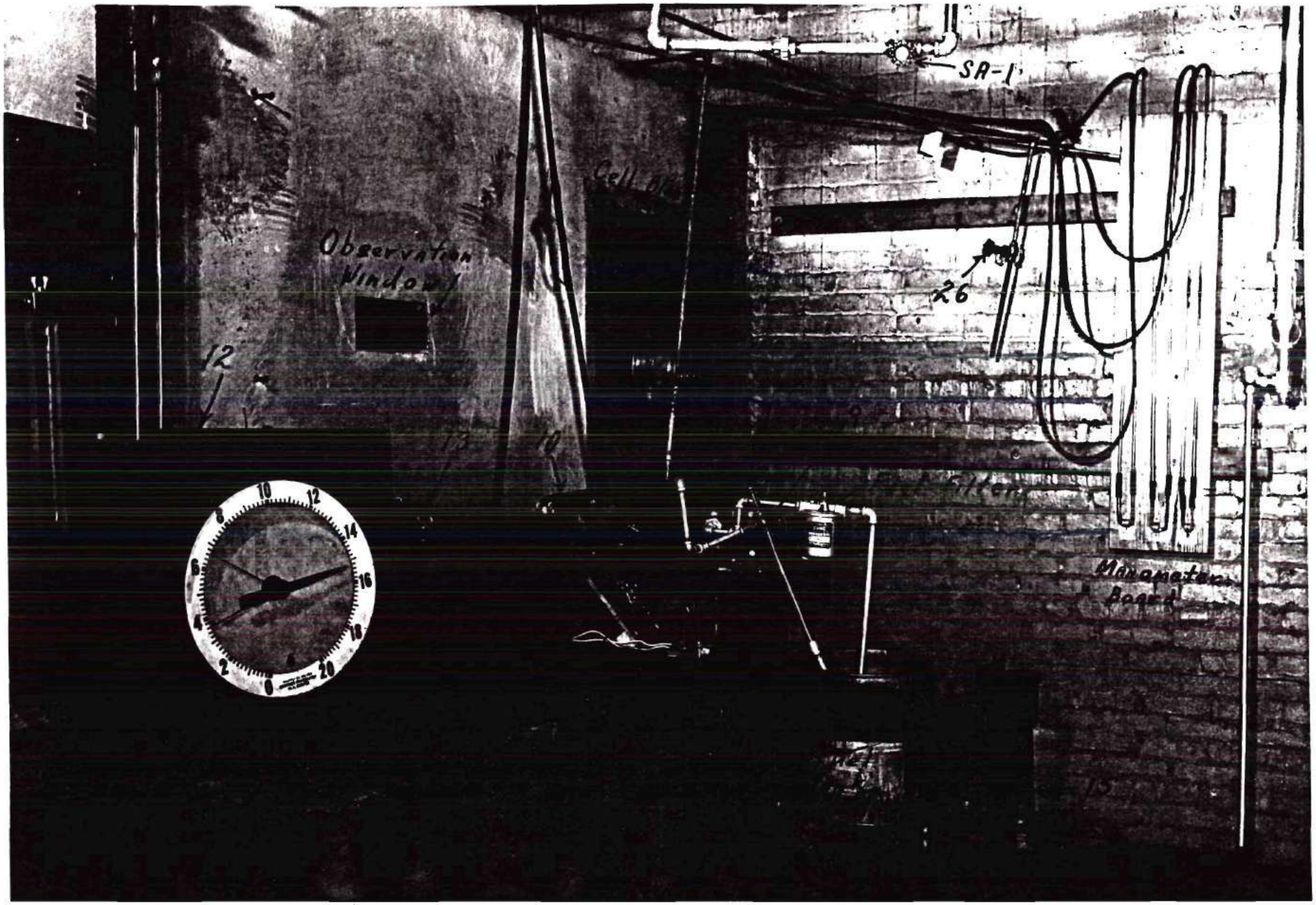
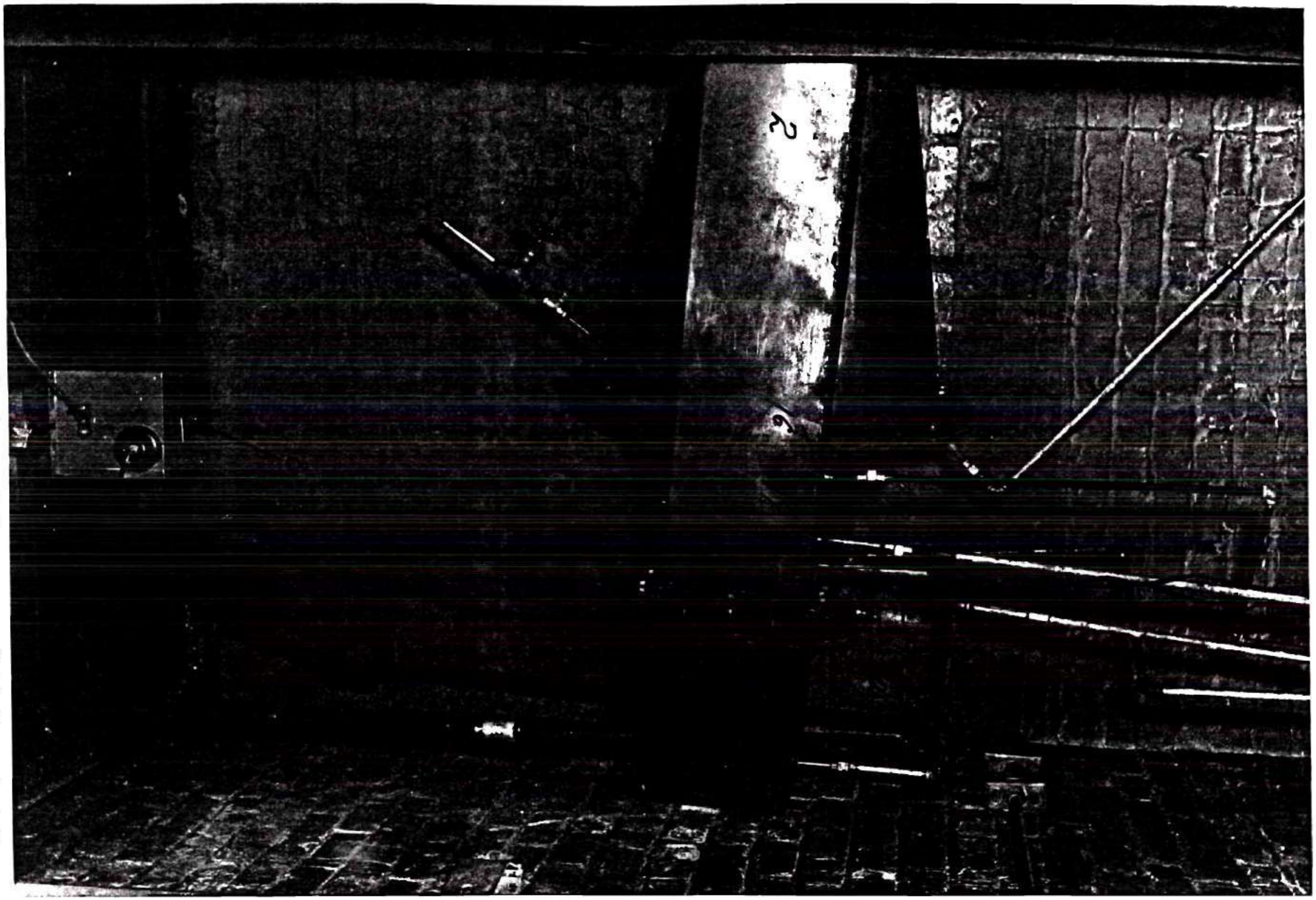


Figure 6 - View of Instruments and Auxiliaries

LEGEND FOR FIGURE 6

Part Number	Description of Part
9	Fuel Pump
10	Fuel Pump Driving Motor
12	Potentiometer Pyrometer
13	Selector Switch
14	Tachometer
15	Beam Scale
26	Turbine Inlet Impact Pressure By-Pass Valve
S-1	Transformer Switch
S-2	Bearing Blower Motor Switch
FP-1	Fuel Pressure Control Valve
FP-2	Fuel Shut-Off Valve
SA-1	Starting Air Shut-Off Valve



LEGEND FOR FIGURE 7

Part Number	Description of Part
2	Compressor Duct
3	Combustion Chamber
8	Transformer
16	Compressor Duct Static Pressure Tap
17	Compressor Duct Impact Pressure Tap
18	Compressor Duct Thermocouple
19	Combustion Chamber Inlet Static Pressure Tap
20	Combustion Chamber Inlet Thermo-couple
FP-3	Excess Fuel Drain Valve

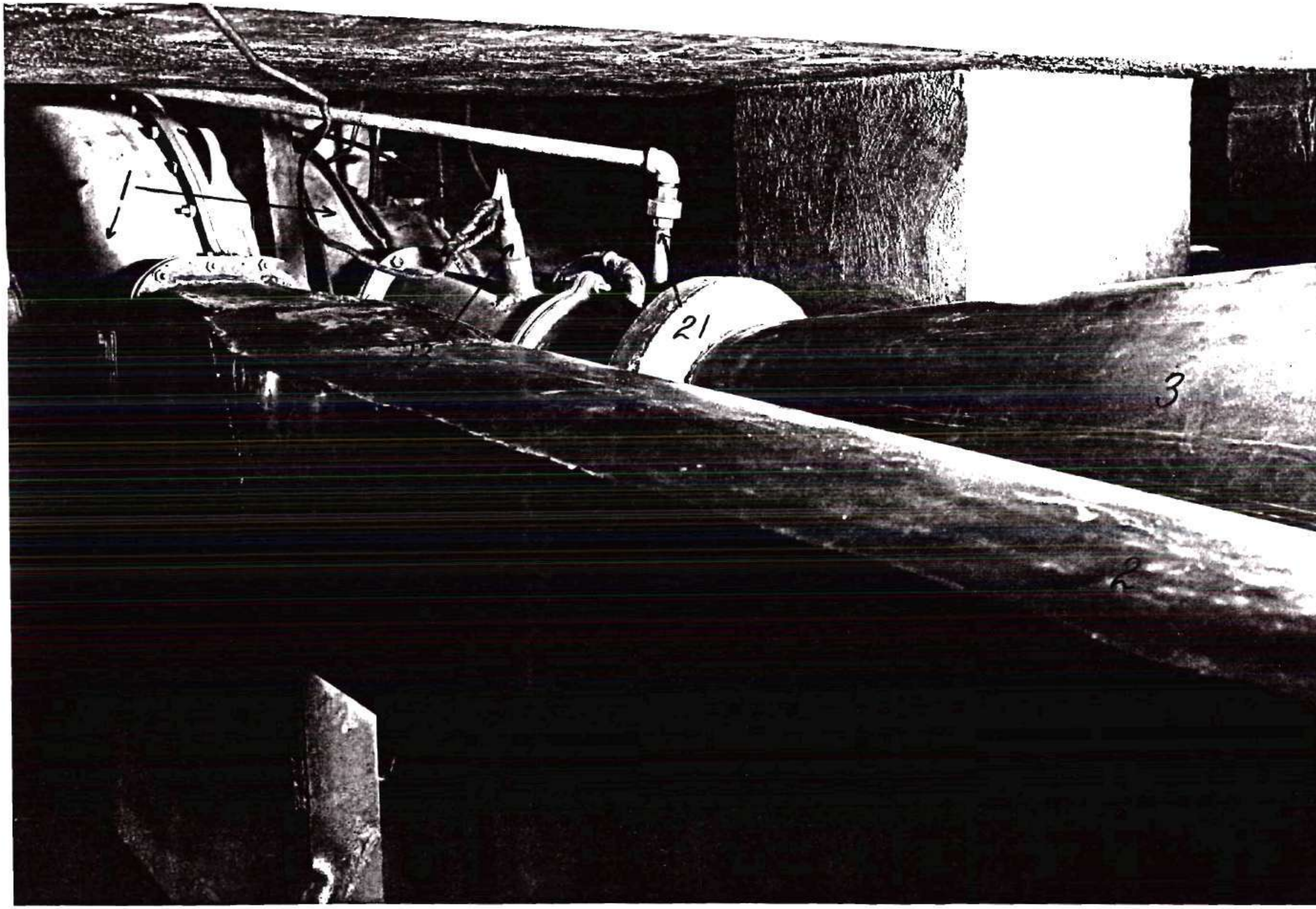


Figure 8 - View of Combustion Chamber, Compressor Duct, and Turbosupercharger from Inside Cell Block

LEGEND FOR FIGURE 8

Part Number	Description of Part
1	Turbosupercharger
2	Compressor Duct
3	Combustion Chamber
21	Turbine Inlet Static Pressure Tap
23	Vertical Turbine Inlet Thermocouple

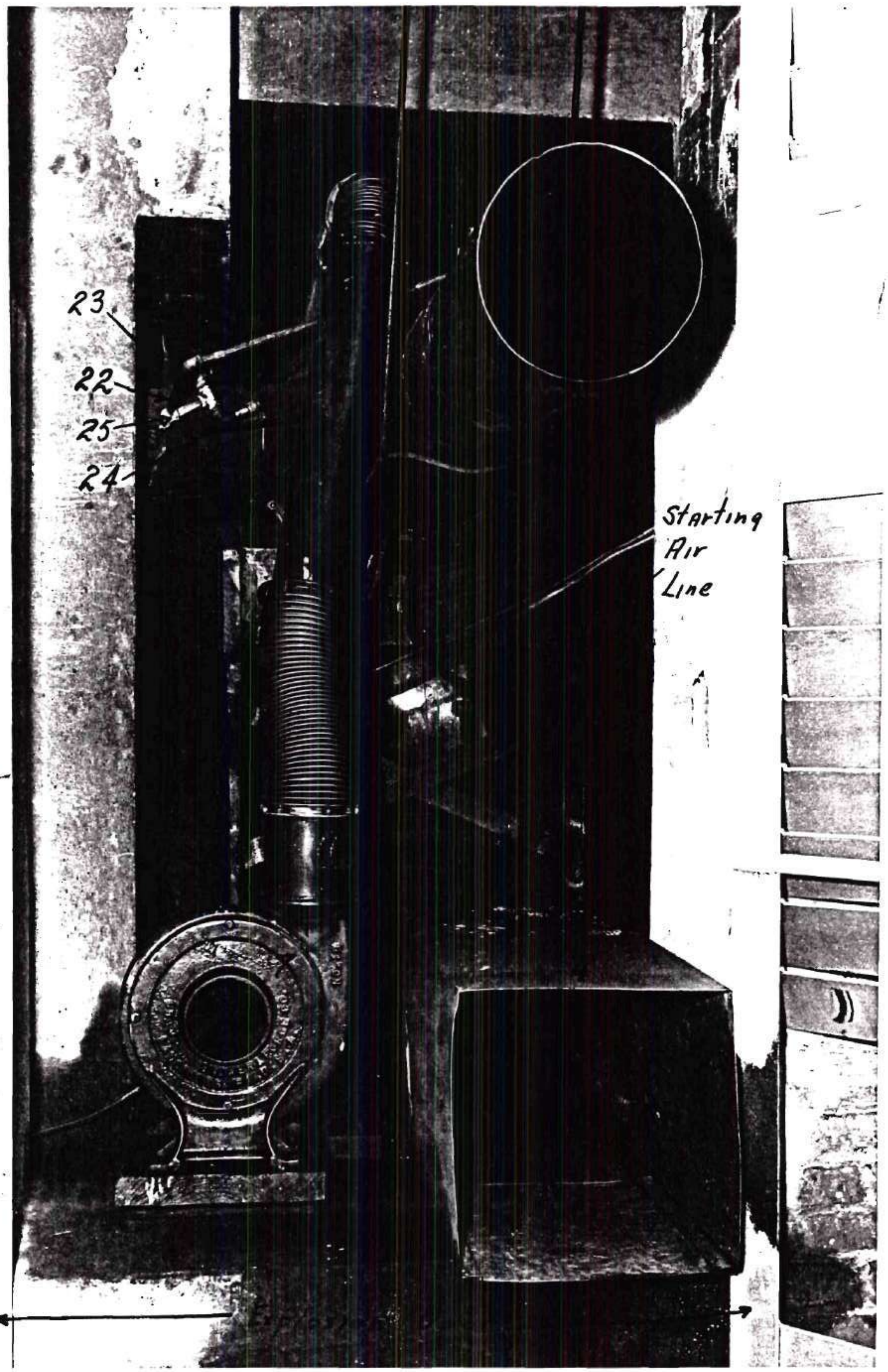


Figure [illegible]. View through Explosion Door

LEGEND FOR FIGURE 9

Part Number	Description of Part
4	Intake Air Duct
5	Exhaust Duct
6	Bearing Blower Motor
7	Bearing Blower
22	Turbine Inlet Impact Pressure Tap
23	Vertical Turbine Inlet Thermocouple
24	Horizontal Turbine Inlet Thermocouple
25	Turbine Inlet Aspirating Thermocouple

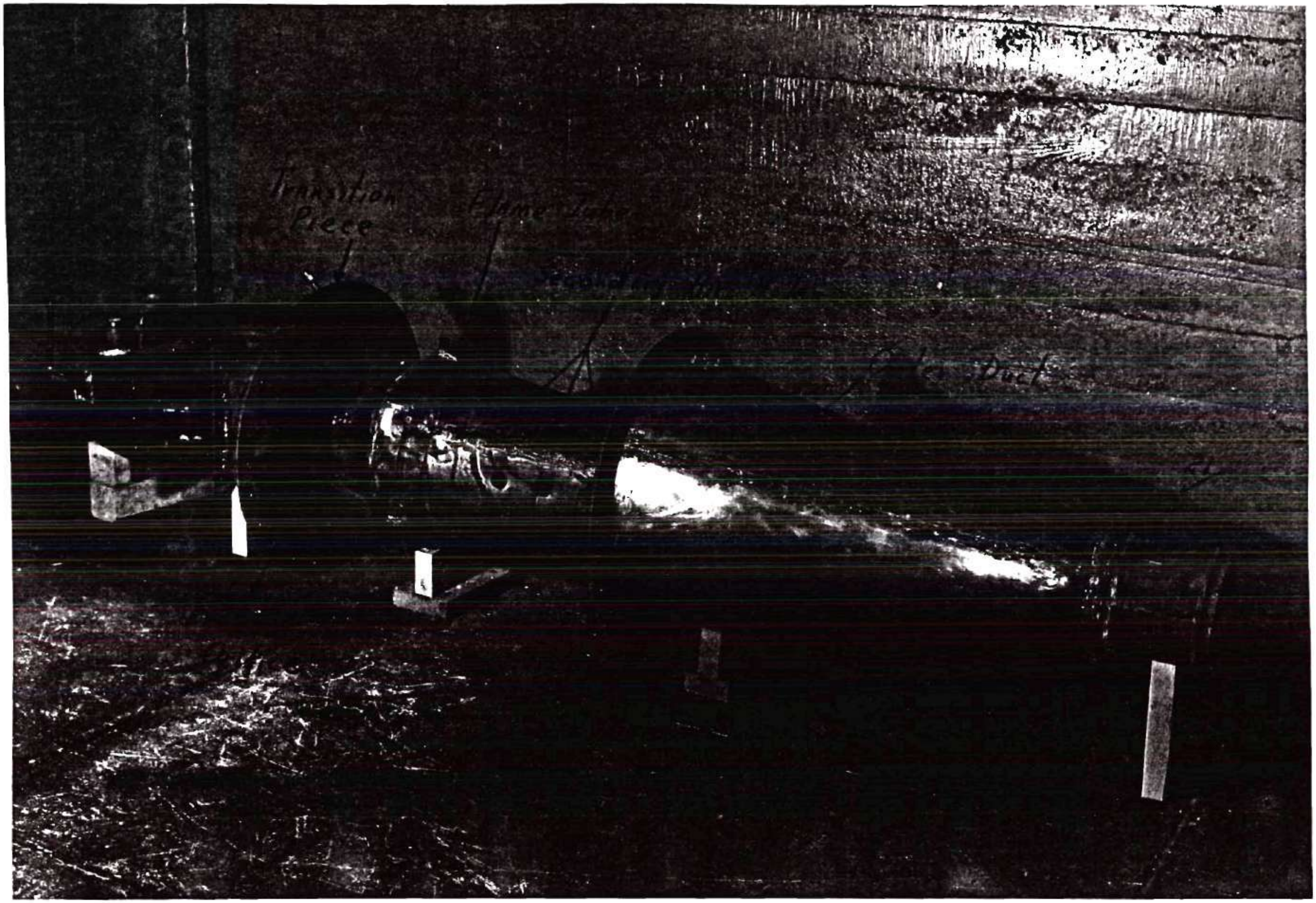


Figure 10 - Explosion View of Combustion Chamber

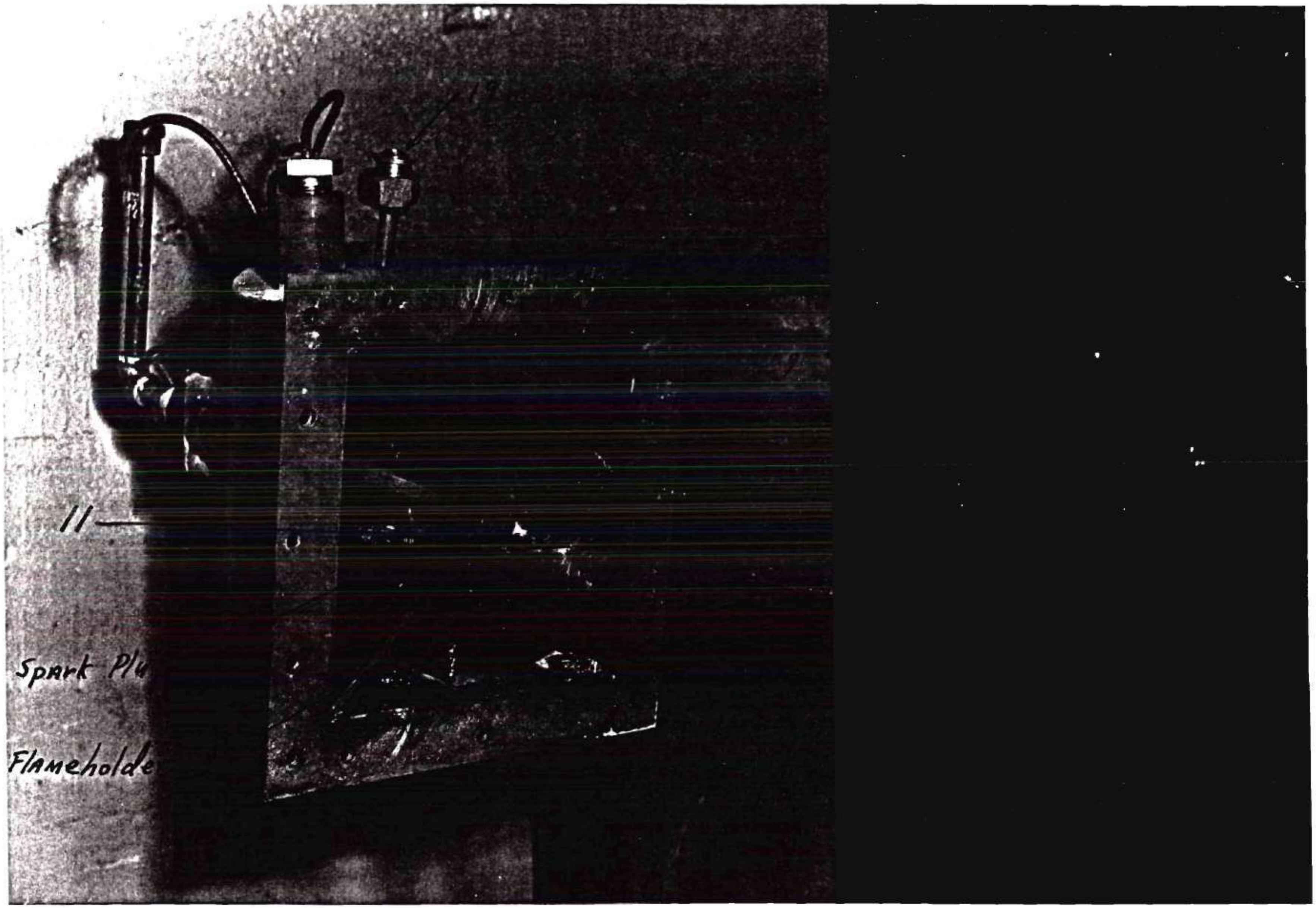


Figure 11 - Close-up of Fuel Nozzle, Flame Holder and Spark Plug

LEGEND FOR FIGURE 11

Part Number	Description of Part
11	Fuel Nozzle
19	Combustion Chamber Inlet Static Pressure Tap
20	Combustion Chamber Inlet Thermocouple Well

APPENDIX II

OPERATING INSTRUCTIONS

The following are steps which the writer feels should be followed when starting and running the bootstrap. It is suggested that they be strictly adhered to until such time that the operator has acquired sufficient familiarity with the unit to develop his own technique.

1. Switches no.29 and 30 on the laboratory switch board should be on. These close the circuits to the test block lights and auxiliaries.

2. Drain excess fuel from flame tube by opening valve no.FP-3 until flow has stopped. Close valve.

3. Check to see that the oil tank is at least half full. If not, add S.A.E. 20 oil.

4. Check hose connections from bearing blower to turbine cooling cap.

5. Throw switch no.5-2. This closes circuit to bearing blower motor.

6. Close and lock steel door to cell block.

7. Close explosion door and check for foreign matter in intake duct.

8. Connect fuel pump and potentiometer to 110 volt supply outside the cell block.

9. Close valve no.FP-2 (fuel shut-off valve).

10. Start fuel pump motor and adjust fuel pressure with valve no.FP-1 (fuel pressure control valve) until pressure gage reads 1000 psi to insure that the fuel pump and driving motor are in proper running shape. Reduce pressure to zero.

11. Standardize potentiometer. If this is not done, the temperature indicated by the instrument are meaningless.

12. Throw thermocouple switch no.4 to the on position. The potentiometer now reads turbine inlet temperature.

13. Turn valve no.SA-1 (starting-air shut-off valve) to full open position. When tachometer reads between 2000-2500 rpm proceed.

14. Adjust fuel pressure to 20 psi.

15. Throw switch no.S-1 to on position. This supplies power to the 110-15,000 volt transformer.

16. Open valve no.FP-2.

17. If temperature and speed do not begin to rise within 30 seconds, shut off valve no.FP-2 and switch no.S-1 but leave starting air on so as to purge the combustion chamber of fuel particles. After 3 minutes repeat nos.15 and 16.

18. If unit does not fire after second try, shut down; unit should be disassembled and trouble corrected.

19. If unit fires, wait until tachometer indicates 6000 rpm and shut off starting air supply.

20. Do not shut unit down because of noise caused by a low frequency vibration. It is caused by the flame tube, and will

stop as soon as it has expanded and wedged itself against the turbine end of the outer duct.

21. Throw switch no.S-2 to off position. The spark is not required after the flame front is established.

22. The unit is now self-operating. All further speed control is accomplished by adjusting pressure control valve. Never exceed pressure of 1200 psi, otherwise fuel pump driving motor will stall out.

23. Never allow turbine inlet temperature to exceed 1600°F.

24. Never allow speed to exceed 20,000 rpm.

25. In case of emergency or when shutting down unit, reduce fuel pressure to zero and close fuel shut off valve. Reducing fuel pressure to zero is not enough to insure fuel stoppage, since nozzle drips when fuel shut-off valve is not closed.

26. Shut off all auxiliaries.